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DESIGN CONSIDERATIONS IN MECHANICAL FACE SEALS FOR IMPROVED PERFORMANCE
I. BASIC CONFIGURATIONS

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TECHNICAL PAPER to be presented at the
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ABSTRACT

Basic assembly configurations of the mechanical face seal are described and some advantages associated with each are listed. The various forms of seal components (the primary seal, secondary seal, etc.) are illustrated, and functions pointed out. The technique of seal pressure balancing and its application is described; and the concept of the PV factor, its different forms and limitations are discussed. Brief attention is given to seal lubrication since it is covered in detail in a companion paper. Finally, the operating conditions for various applications of low pressure seals (aircraft transmissions) are listed, and the seal failure mode of a particular application is discussed.

INTRODUCTION

In a highly technical society the fluids which must be sealed range from water (automobiles, reactors, submarines) and oil, to liquid oxygen and toxic chemicals. The mechanical face seals developed for these applications have many diverse forms, from the low cost automotive water pump seals to the very sophisticated seals for liquid oxygen turbopumps such as are used on the space shuttle. It is well recognized that seals can have a significant cost impact in regards to maintenance and downtime. An additional consideration that has received recent attention is the health hazard. It is now realized that personnel exposure to even low levels of some substances can have serious health consequences; often the health damage does not become manifest until late in life.

In regard to chemicals which may be a health hazard, ref. (1) points out that one source of worker exposure comes from leaks in valves and pumps, and states that prevention of a health hazard (occupational disease) is highly desirable since it tends to be chronic, untreatable, and fatal; and may go unnoticed if it does not result in an unusual group of clinical manifestations (the case of a rare neoplasm of the liver found in workers exposed to vinyl chloride is an example). An additional point in regard to the seriousness of industrial carcinogens is made by ref. (2), in which it is stated that, unlike cigarettes smoking where the risk of lung cancer in ex-smokers is only slightly above that of lifetime nonsmokers, industrial carcinogens cause an excess risk which does not appear to drop after exposure ends. Specifically, ref. (2) lists numerous substances which appear to be associated with excessive lung cancer mortality, among these are polycyclic aromatic hydrocarbons, bis chloromethyl ether (BCME) and chloromethyl methyl ether (CMME). In this regard, the Occupational Safety and Health Administration (OSHA) has developed exposure limits for over 300 substances and additional substances will probably be added as research information is developed.

In general, face seal application is largely empirical, being based on past experience. However, current technical needs (e.g., breeder reactors) and the previously cited occupational disease considerations have placed an additional burden on a technology, which seems to some, to have been barely adequate in certain applications.

The objective of this study is to review the basic seal configurations and design considerations; a companion paper, ref. (3), summarizes the current thinking on seal lubrication, which is a major factor in improved life. In this paper, discussion is limited to the conventional face seal. High-performance special-purpose seals that employ thrust bearing geometry machined into the primary-seal surfaces (hydrodynamic seals) are not addressed; an introduction to these can be found in ref. (4).

DISCUSSION

Applications

The mechanical seal is recognized as one of a family of devices used in the general area of fluid sealing of rotating shafts; this general area of sealing is represented by the pyramid in fig. 1. At the bottom, or base, are packings which were the earliest and still the most frequently used solution to sealing problems. Moving up the pyramid, there is an increasing complexity of sealing devices including O-rings, lip seals and mechanical seals, beginning with the simpler versions used in the appliance and automotive industries. Moving up the pyramid further there are aircraft, marine, and chemical process industry applications. Finally, at the peak there are sophisticated aircraft and nuclear requirements, and rocket component sealing systems.

Arrangements of Seal Components

One conventional arrangement of seal-face components is shown in fig. 2. The primary-ring has axial flexibility, usually contains the primary seal diameters, and also has angular flexibility. The axial flexibility allows accommodation of axial displacements that arise for various reasons (e.g., face runout, thermal growth differences and tolerance variations). The secondary ring can be of various designs, such as elastomeric "O"-rings, bellows, or piston rings.

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Nomenclature Applying to Single Parts

Seat or mating ring	part having a primary-seal face and being mechanically constrained with respect to axial motion (e.g., attached to the shaft or housing)
Primary-ring	ring having a primary-seal face and flexibility in axial and angular modes
Secondary-ring	part (O-ring, piston ring) having secondary-seal surfaces that mate to the secondary-seal surfaces of the primary-ring and housing

Nomenclature Applying to an Assembly of Parts

Primary-seal	seal formed by the primary-seal faces of the seat and primary-ring; relative rotation occurs between these seal faces
Secondary-seal	seal formed by the secondary-seal surfaces of the secondary-ring and the primary-ring (in the case of a bellows, the secondary-seal is the bellows itself)
Film thickness, h	distance between primary-seal faces (h may vary with radial or circumferential position and with time)

Basic Configurations

Basically, a mechanical face seal (fig. 2) prevents leakage of a fluid along a rotating shaft which passes through a housing or pressure vessel. Sealing is accomplished by a stationary (nonrotating) primary-seal ring that bears against the face of a mating ring (seat) mounted on the shaft. Axial loading, from the sealed pressure and the springs, maintains the "contact" between the primary-ring and the mating ring. Sealing occurs at the primary-seal faces.

The mechanical face seal was illustrated (fig. 2) with a nonrotating primary-ring and a rotating mating ring (seat) attached to the shaft. It is apparent that these functions could be reversed; therefore, considering that the pressure could be either at the outside diameter of the primary-ring or at the inside diameter, four basic seal assembly configurations can be identified; these are, a rotating primary-ring with the sealed liquid at the O.D. or the I.D., and a nonrotating primary-ring with the sealed liquid at the O.D. or the I.D. (see figs. 2 and 3). Occasionally counter-rotating and differential speed applications are encountered where both the primary-ring and the mating ring are rotating.

The most common configuration, especially in pumps, is the rotating primary-ring configuration (fig. 3); the principle advantage of this configuration lies in the relative ease of attachment to straight shaft sections without affecting the seal face alinement. The primary-ring assembly is easily attached to the shaft by set screws, friction fits, or similar means which assemble directly on a smooth shaft diameter.

Nonrotating primary-ring configurations (fig. 2) is often found in applications where space is limited, or where the rotating speeds are high. The principle advantage of the nonrotating primary-ring is the absence of centrifugal forces on both the primary and secondary rings.

The rotating primary-ring with the sealed liquid at the O.D. has a number of advantages as compared to the rotating primary-ring with the sealed liquid at the I.D. These are:

1. Centrifugal force tends to retard leakage
2. Centrifugal force tends to centrifuge solid particles away from the primary-seal and springs, thus there is a tendency to be self cleaning
3. The sealed pressure tends to put the seal rings into compression

On the other hand the rotating primary-ring I.D. configuration has advantages in that it is easier to install; and in sealing corrosive fluids, the seal can be designed so that metal parts do not come into contact with the corrosive fluid.

Tandem Seal Arrangements

For dangerous liquid, seals are often used in tandem (fig. 4) with a barrier liquid between the seals. The barrier liquid may be pressurized to a level slightly above that of the sealed process liquid in order to preclude any leakage of the sealed liquid; in other designs, the barrier liquid is held at a pressure equal to or even slightly below that of the process liquid. Selection of the barrier liquid pressure depends on restrictions regarding leakage rates of the barrier liquid into the process liquid, or on the process liquid leak rate into the barrier liquid.

It should be noted that holding the barrier liquid pressure above that of the sealed process liquid is no absolute guarantee against leakage of the process liquid into the barrier liquid (and then from leaking into the atmosphere). In this regard ref. (5) cites the experience with deep well irrigation pumps in which water leaked across the seal into the bearing lubricant side even though the lubricant was at a higher pressure than the water. Evidently the seal acts as a small pump, transferring the fluid from a lower pressure to a higher pressure. The mechanism responsible for this pumping action was investigated by several researchers (refs. (6) and (7)) and one explanation is that certain combinations of parallel misalinement and primary-seal surface waviness result in a pumping effect against the higher pressure. (Surface waviness and parallel misalinement are discussed in a later portion of this paper.) Obviously, if the process liquid is very toxic, and very low levels of atmosphere contamination must be maintained, then the possibility of process liquid transport across a seal into a region of higher pressure is a major concern.

It is apparent that many different combinations of tandem seal configurations can be constructed from the four basic seals (rotating primary-ring and nonrotating primary-ring, either of which can be O.D. or I.D. pressurized). The advantages and disadvantages of each involve cooling, leakage, safety, and cost considerations which are outside the scope of this paper. Several of many possible configurations are shown in fig. 4; and it should be noted that tandem seals are often used with heat exchangers to maintain the seals at a desired temperature level.

The Primary-Seal

The primary-seal performs two functions; these are, (a) effectively retains the pressurized fluid and (b) acts as a thrust bearing using the sealed liquid as a lubricant. (There is a very close parallel between seal face operation and thrust bearing operation.) When selecting materials for use as seal faces, a general rule is that all good face seal materials must be good bearing materials, but not all good bearing materials will necessarily make good seal faces. Carbon-graphite materials of various types,

used in combination with some other compatible hard material, are used as face materials in the vast majority of mechanical seal applications. Table I lists a number of materials used for seal faces and the typical environments and associated material combinations that are frequently used.

In seals of all types, a basic difficulty is found in the fact that the seal lubricating film is very thin (in the range of 1 micron or less) and therefore, very small surface irregularities, elastic displacements (thermal bumps, etc.) and face runout motions have a dramatic effect on the lubrication of the seal. Thus the primary-seal cannot, in general, be visualized as two perfectly flat and parallel surfaces; some possible geometries are illustrated in fig. 5. There have been many hypotheses put forth to explain the mechanism (or mechanisms) responsible for the development of the lubricating film pressure that acts to separate the primary-seal faces. These hypotheses include the following: surface angular misalinement, surface waviness, surface asperities, vaporization of the fluid film, axial vibration, and thermal deformation; current work is reviewed in the companion paper (ref. (3)).

With reference to the possible primary-seal geometries of fig. 5, waviness (geometry a) and angular misalinement (geometry b) are most likely sources of hydrodynamic pressures that contribute to face-seal lubrication. Coning (geometry c) affects the hydrostatic force balance (not significant in low-pressure seals) and this is also discussed in ref. (3). Externally imposed axial vibration (geometry d) can produce useful squeeze-film pressures, but it is judged not to be significant because of the high damping that is associated with viscous fluids and secondary-seal friction. Parallel misalinement and shaft whirl (geometries e and f) do not produce significant fluid-film pressures directly but can affect the transport of fluid into and out of the primary seal. In this regard, when the primary ring rotates (nonrotating seat), parallel misalinement does not influence seal operation (leakage, fluid transport, etc.), since in rotating, all the points on the surfaces of the primary-seal remain within the primary-seal. However, when the seat rotates and parallel misalinement is present, some points on the seat surface enter and leave the primary-seal during each revolution. This radial velocity component can affect both leakage and lubrication.

The Secondary-Seal

The secondary-seal elements may be divided into three basic types:

(a) Compression packings
(b) Automatic packings
(c) Bellows diaphragms

Compression packing was one of the earliest used in mechanical seals (see fig. 6). The principle feature of the compression packing system is that it utilizes a mechanical load on the packing. The typical arrangement, shown in fig. 6(a), consists of a small packing box, one or two rings of compression-type packing and a gland follower actuated by the same spring load used to load the seal faces. Dynamic packing rings used with this construction include a variety of materials, some of which are molded synthetic elastomers, asbestos, and synthetic binders, carbon-base packings and plastic materials, the most common of which would be PTFE.

The term "automatic packings" apply to all of those packing rings and devices that are self energized by the sealed pressure and do not normally require an auxiliary mechanical load to maintain sealing contact. Figure 6(b) illustrates a typical mechanical seal using an automatic packing ring. Some of the variations commonly used in mechanical seals include O-rings, U-ring, and piston rings. Seals using piston rings as the secondary seals generally use conventional piston rings of various designs. Figure 6(c) illustrates a typical high temperature piston ring assembly. These are usually special purpose seals operating in severe temperature environments beyond the capabilities of the elastomers described previously.

The third category of secondary seals, bellows and diaphragms, can be subdivided into two groups:

(a) Elastomer bellows and diaphragms
(b) Metal bellows and diaphragms

Figure 6(d) illustrates the typical elastomeric bellows seal assembly. Sliding packing surfaces and the friction associated with them are eliminated. Within temperature and pressure limits and compatibility with the media, the elastomeric bellows is one of the most widely used of all mechanical seals.

Figure 6(e) illustrates a formed metal bellows arrangement. This was the original construction used for all metal seal construction.

Figure 6(f) is a later development illustrating a welded metal bellows seal assembly usually found in applications involving high temperatures or highly reactive media where synthetic elastomers are incompatible with the media. The welded nesting bellows requires less space, has softer spring rates, wider operating ranges and has higher pressure capabilities than the formed bellows.

Because of the all metal construction, these welded metal bellows, fabricated from corrosion resistant alloys, will be found in high temperature service and in various types of corrosive media. Several additional advantages of this construction are the elimination of sliding interfaces and, therefore, the elimination of abrasion and fretting corrosion. The metal bellows also eliminates the need for separate springs since the bellows serves both the sealing function and the face loading function.

In general, the secondary-seal has a dramatic effect on the performance of the seal, but this is not well understood. Also the frictional force of the secondary-seal can change substantially with time, and this alters the seal's performance. In extreme cases, the secondary seal friction increases to a point were the seal sticks open; an example is "O" ring swelling due to lack of compatibility with the sealed fluid.

Loading Elements

Figure 7 illustrates the various springs that may be found in conventional mechanical seals, starting with the single spring (fig. 7(a)), the multiple spring (fig. 7(b)), wave spring (fig. 7(c)), belville springs (fig. 7(d)) and bellows (fig. 7(e)).

The single spring has the advantage that it is a single component having a relatively large wire cross section with less susceptibility to degradation through corrosion. The disadvantages are that a new spring is required for every shaft size, the single spring is a long assembly, and centrifugal force may affect the coils at high speed causing them to open up.

Multiple springs permit the use of a standardized spring for a variety of sizes. Loads are varied by simply varying the number of springs in the assembly. Multiple springs will operate at a relatively shorter operating length than the single spring design. Disadvantages are described as the number of added parts

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in an assembly, and the potentially greater effect on spring loads for a given corrosion rate.

Wave springs (fig. 7(c)) are used in seals designed for very small axial spaces. Their advantages are principally lightweight and limited space requirements. The disadvantages can be described as a low available working length, degradation of the spring loading due to corrosion of the thin materials, and the probability of wear under conditions of high vibration or excessive movement.

Belville springs, which are made up of a series of disk washers (fig. 7(d)), have been used in special seal construction. The advantages of this technique is that rates can be changed by simply adding or subtracting washers. Projected loads from belville springs will be generally more diametrically uniform than any of the above springs. The disadvantages of the belville spring is its initial cost due to tooling required and relatively high spring rates that must be reduced by a multiple number of convolutions. There is also a greater likelihood of fouling the spring system through deposits of foreign material on the inside convolutions of the belville plates.

Figure 7(e) illustrates a formed metal bellows arrangement which was the original construction used to obtain an all metal seal assembly. A later development is the welded nesting bellows; a seal assembly with bellows is usually found in applications involving high temperatures or highly reactive media where synthetic elastomers would be incompatible with the sealed liquid. The welded bellows requires less space, has lower spring rates, wider operating ranges and has higher pressure capabilities than the formed bellows. Finally, a seal using magnets instead of springs is shown in fig. 7(f); this provides a very compact assembly.

Pressure Balance

Pressure balance is defined as the ratio of two areas. Referring to fig. 8 (a typical balanced seal) the primary-seal area, which is the area bounded by the outer and inner diameters of the sealing face, have been designated A_f. The closing force area, which is the area subjected to the sealed pressure and bounded by the secondary seal balance diameter and a primary face diameter has been designated A_p. The pressure balance ratio is the value of A_p over A_f. Expressed in terms of the face diameters, the pressure balance is derived as shown in fig. 8.

In simpler terms, it can be stated that the unit hydraulic loading transmitted to the seal face of a balanced seal is some value less than the pressure sealed. The range of pressure balance used for balance seals will generally vary over a range from 60 to 85 percent dependent upon the characteristics of the application. The great majority of seals are balanced in the range of 70 to 75 percent.

When sealing an incompressible liquid and operating on a lubricating film, the pressure drop across the sealing face as calculated for incompressible flow is a straightline function.

The average film pressure will, therefore, equal 50 percent of the pressure sealed. If this factor is equated against the seal's hydraulic balance, it will be seen that the residual load available to maintain the faces in contact on a 70 percent balance seal would be theoretically 20 percent of the pressure sealed, which is the net load available to maintain the faces in contact.

If we are considering a gas as the fluid sealed, the flow characteristics would be that of a compressible fluid. The pressure profile across the sealing face would occur in the form of a curve and the required balancing characteristics would be adjusted accordingly.

The unbalanced seal (fig. 9), is one in which the full value of the pressure sealed is impressed upon the sealing faces. Unbalanced seals will generally be used for low pressure service and balanced seals will be applied to high pressure applications.

Arbitrary recommendations ranging from 100 to 200 psi are usually established for the operating pressure limits between balanced and unbalanced seals. In practice, there is no fixed value for this decision point and all of the operating conditions as well as the characteristics of the media sealed may influence the decision. Unbalanced seals are operating successfully in service that range as high as 1200 psi involving special operating conditions and duty cycles.

Figure 10 illustrates some variations that may be applied to the primary seal face detail. A wide face and a narrow face are illustrated. The hydraulic balance of either face can be identical. Nevertheless, the two faces may perform differently. Torque requirements and heat generation will be a function of the total load and hence a function of the total face area. The narrower face will result in lower torque, lower heat generation for a given set of operating conditions. This would imply that all faces should be made as narrow as possible. This must be tempered by some practical considerations. The most important point is that the lapped faces are flat to the limits of a manufacturing tolerance. When subjected to mechanical and pressure loadings, further variations in face flatness will occur. Since the film pressure is no longer uniform, contact will take place at some points and the load will be distributed between some fluid film and some contact area. For a given variation in flatness, the shorter leakage path of the narrow face will result in higher leakage rates than the wide face seal arrangement.

There are no fixed boundary lines or hard-and-fast rules for face width. In general, narrower faces are used for high speed where the lower torque and heat generation may be critical to the performance. Wider faces will be found on high pressure, low speed applications where lower stresses will provide lower distortion, thus maintaining the flatness characteristics, and this promotes lower leakage rates.

PV Factor

An arbitrary mathematical relationship, often used to evaluate the severity of an application, is the PV value which is the product of the pressure expressed in psi times the velocity expressed in feet per minute.

The resultant PV value is a factor that is useful for establishing a relative degree of severity of an application or for establishing a rough comparison of performance between several applications operating under differing operating conditions. It has developed through informal usage by the seal manufacturers, the seal users, and material suppliers. There are several variations in the calculations that have evolved and it is necessary to be sure that the approach is uniform before comparable relationships can be established and inferences made from these values.

Figure 11, formula 1, is the simplest version as it is the simple product of the pressure and velocity. In this calculation, there is no consideration given to film pressure differentials across the face or geometric characteristics. It is often used to identify the test parameters in face material testing which is conducted with a straight mechanical load on the sealing faces and with no pressure differential across the

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contacting area.

Formula 2 uses the balance factor to reduce the pressure sealed to the unit loading the seal faces actually experience. In the case of an unbalanced seal, the balanced factor is 100 percent; therefore, the calculated PV value would be the same for formula 1 or formula 2. Only the balanced seal would show a proportionately reduced PV value.

Formula 3 is a further modification that has been used by some seal engineers. In this case, the unit loading is further reduced by subtracting the pressure exerted by the lubricating film between the faces. We have previously indicated that an incompressible fluid has a substantially straight line pressure drop and that the average pressure is one-half of the differential. This value is then subtracted from the gross hydraulic load resulting in a net differential hydraulic load which is used for the PV calculation.

The concept of the PV value is an imperfect relationship but, if used in the proper context, it can be a useful reference tool. Of the three versions, formula 2 provides the most logical and realistic comparison; but, the most important concern is to know the basis of the computation when making comparisons of face material performances.

In applications where the pressure differential is very low, such as in helicopter transmission seals in which the pressure differential may be zero, the spring loading magnitude is critical. In these cases the PV factor is calculated by using the contact pressure produced by the springs on the primary-seal area. Examples of PV factors used in applications sealing a gear box lubricants are given in Table II; the first 9 entries are helicopter transmission applications and the data was given by ref. (8). The last two entries in Table II are reported problem areas. Comparison of the PV factors of the last two entries with the first nine does not suggest operation outside the bounds of the state-of-the-art; this illustrates that the PV factor is not an absolute guide for prediction of successful operation.

Figures 12 through 14 depict the type of surface distress that is sometimes found in these types of very low pressure seals. The seal was operated under the conditions listed in line 10 of Table II, these conditions are:

2089 M per min (6870 ft/min) sliding speed; 7.8 N/cm^2 (11.36 psi) face pressure due to spring load and 78,043 (ft/min x psi) PV factor. Instead of a face contact pressure some seal engineers use closing force per unit length of the circumference. On this basis the load due to the springs for the application in line 10 Table II is 2.14 N/cm or 1.22 lbf/in.; and briefly stated, for the seal to operate successfully the primary-seal face must act as an oil lubricated thrust bearing and produce a load capacity equivalent to 2.14 N/cm of seal circumference. However, as indicated in figures 12 through 14, heavy surface distress occurred because of lack of adequate lubricating (load support) film. The seal shows severe heat checking (fig. 12) with temperatures probably reaching over 811 K (1000° F) as indicated by the soft pearlite produced at local hot spots (fig. 13). The wear profile of the rotating seat matched that of the carbon ring and this is shown in fig. 14.

Seal Lubrication

The engineering of seals can involve fluid mechanics, heat transfer, lubrication theory, thermodynamic, chemistry, physics, metallurgy and dynamics, to mention a few of the most frequent areas of concern. Seal problems may consist of superposition of effects which can be interrelated. Usually, each effect can be analyzed by itself, but then integrated effects must be evaluated for a complete analysis of a sealing system. As indicated previously, seals are characterized by surfaces in relative motion separated by a very narrow gap. In order to maintain proper operation, very small differences in the dimension of seal parts must be maintained. Deformations in geometry due to imposed thermal gradients, frictional heating, pressure and mechanical and inertial forces must be held to a minimum. In most cases, any deformations must be no more than microvalues.

Seals operate in many lubrication regimes, depending upon the type of seal, sealed fluid, the application and related characteristics.

Figure 15 illustrates the various seal lubricating regimes that can exist (ref. (8)). In a companion paper (ref. (3)) seal lubrication is discussed in detail; the current theories are reviewed and seals are classified according to the lubrication operation mode.

CONCLUDING REMARKS

The seal descriptions in this paper show that mechanical face seals appear in many diverse forms and that a basic difficulty of all is the very small thickness of the lubricating film (on the order of 1 micron). And since the film is very thin, very small irregularities or motions of the primary-seal can have a dramatic effect on the performance of the seal. (For a detailed discussion on seal lubrication the reader is referred to ref. (3).)

Generally, seal lubrication is not well understood and this is evident when compared to bearing lubrication in which performance can be closely predicted; in seals, prediction of performance (leakage or life) is often not possible. The importance of the secondary seal is stressed, in general, data is lacking in regard to secondary seal friction which must be offset by the spring load or by seal pressure balance; here current practice depends on a vast amount of experience. Finally, the limitations of the PV factor should be kept in mind, in this regard, insights provided by current thinking on lubrication can help guide the application of seals.

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TABLE I. - END FACE SEAL MATERIALS AND ENVIRONMENT COMBINATIONS

Environment	Seal nose material	Shoulder material
Water	Carbon graphite (also carbon graphite containing various metals - copper, lead, babbit, etc.)	Bronze Ni-resist Nickel cast iron (for constant operation only) Ceramic Stellite (hard facing on 316 stainless or any stabilized stainless) Tungsten carbide Malcomized 316-stainless Carbon-filled PTFE Glass-filled PTFE Chrome plate on various parent materials (must be thick enough) Ceramic Facing on stainless
	Tungsten carbide	Tungsten carbide
	Carbon containing various metals	Stainless-steel (series 400, hardened to Rockwell C-50 or higher)
Caustics	Carbon graphite	Carbon-filled PTFE Stellite-faced stainless steel
	Carbon graphite (nonmetallic)	Hard-faced 316-stainless steel
	Carbon graphite (metallic, for dilute solutions)	Stellite faced stainless steel
Salt Solution	Carbon graphite	Stainless steel Ceramic Monel
	Ceramic	Ceramic faced stainless Ceramic
	Carbon babbit	Phosphor bronze
Sea Water	Carbon babbit	Aluminum bronze
	Stellite on stainless	Aluminum bronze
	Tungsten carbide	Tungsten carbide
	Bronze	Laminated plastic
	Carbon graphite	Stellite
Acids	Carbon graphite	Hard-faced 316-stainless Carpenter 20 stainless Stellite Chromium boride Ceramic Hastelloy A,B,C Carbon-filled PTFE (for nonoxidizing acids)
	Ceramic	Stellite (attacked by many mineral acids) Glass-filled PTFE (oxidizing acids) PTFE
Gasoline	Carbon graphite	Cast iron Carbon filled PTFE Glass-filled PTFE Ni-resist Nitralloy Ceramic Stellite facing on stainless steel Stainless steel, 400 series
Oil	Carbon graphite	Bronze (for few applications) Ni-resist Cast iron Ceramic Stellite (hard facing on 316-stainless steel, especially for high pressures and high velocity) Tungsten carbide Malcomized 316-stainless Carbon-filled PTFE Glass-filled PTFE Sintered iron or bronze Nitralloy, hardened Tool steel, hardened SAE-1040 steel Stainless steel (400 series hardened to Rockwell C-50. This is general recommendation as 316-stainless is not hardenable)
	Cast iron	Bronze
	Graphite molybdenum	Bronze
Oxidizing Fluids	Boron carbide	Boron carbide
Slurry	Carbon	Ceramic Tungsten carbide
	Bronze	Tungsten carbide
	Tungsten	Tungsten
Heat Transfer Fluids (Dithenyl class)	Carbon graphite	Ni-resist Stellite Ceramics Chrome carbide plate *Tungsten carbide *Ceramic
	Tungsten carbide	*Tungsten carbide *For high temperature capabilities approximately 700° F max.
Gas (Air, CO_2, H_2, He, N_2, O_2)	Carbon graphite	Tool steels Chrome plate Tungsten carbide, plate and solids Chrome carbide plate Ceramics 300-stainless steel 400-stainless steel
	Glass-filled PTFE Carbon filled PTFE (not for H_2 service) (Sodium and fluorine compounds and radioactivity may adversely affect PTFE)	440-C 4140, 4340 Tool steels (hardened) Chrome oxide

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TABLE II. - LOW-PRESSURE FACE SEAL EXPERIENCE

	Application	Seal type	Sealed media	Materials			Face area, cm^2 ($in.^2$)	Seal diameter, mm (in.) (a)	Shaft speed, rpm	Rubbing speed, m/min (ft/min)	Spring load, N (lbf)	Face pressure, N/cm^2 (psi)	Face width, mm, (in.)	PV factor, psi x ft/min	Maximum fluid pressure, N/cm^2 (psi)	Average leakage, cc/hr	Spring load, lb/in. of circumference
				Primary element	Mating surface	Secondary seal											
1	Input shaft	Face	MIL-L-7808 MIL-L-23699	P658RC, CCA 72 or USG 39	440C stainless	Fluoro-elastomer	8.00 (1.2434)	104 (4.115)	13 600	4 460 (14 650)	80 (18)	9.96 (14.47)	2.38 (.094)	212 069	2.76 (4)	5	1.36
2	Output shaft	Face	MIL-L-7808 MIL-L-23699	P658RC, CCA 72 or USG 39	440C stainless	Fluoro-elastomer	4.93 (.7636)	77 (3.037)	6 023	1 460 (4 800)	62.2 (14)	12.67 (18.33)	1.98 (.078)	88 000	3.44 (5)	2	1.43
3	Input shaft	Face	MIL-L-7808 MIL-L-23699	P658RC, CCA 72 or USG 39	440C stainless	Fluoro-elastomer	6.92 (1.0727)	89.5 (3.537)	6 023	1 700 (5 590)	71 (16)	10.30 (14.92)	2.38 (.094)	83 378	1.38 (2)	5	1.40
4	Input shaft	Face	MIL-L-7808 MIL-L-23699	P658RC, CCA 72 or USG 39	440C stainless	Fluoro-elastomer	4.37 (.6777)	55.7 (2.200)	6 023	1 060 (3 480)	57.8 (13)	13.22 (19.18)	2.38 (.094)	66 753	1.38 (2)	1	1.80
5	Input shaft	Face	SATO 35	P658RC, CCA 72 or USG 39	440C stainless	Fluoro-elastomer	7.77 (1.2047)	101 (3.984)	9 000	2 890 (9 500)	75.5 (17)	9.72 (14.11)	2.38 (.094)	134 054	3.44 (5)	4	1.33
6	Accessory	Face	SATO 35	P658RC, CCA 72 or USG 39	440C stainless	Fluoro-elastomer	11.12 (1.7253)	111.5 (4.392)	4 600	1 615 (5 300)	80 (18)	9.35 (13.58)	2.38 (.094)	71 984	1.38 (2)	3	1.28
7	Accessory	Face	SATO 35	P658RC, CCA 72 or USG 39	440C stainless	Fluoro-elastomer	2.71 (.4199)	53 (2.093)	8 000	1 330 (4 380)	57.8 (13)	21.35 (30.96)	1.57 (.062)	135 596	2.07 (3)	1	1.92
8	Clutch APP	Face	MIL-L-7808 MIL-L-21260	USG 39 or P658RC	416 Chrome Plated	Fluoro-elastomer	3.92 (.6072)	68.1 (2.690)	8 200	1 760 (5 780)	26.7 (6)	6.81 (9.88)	1.775 (.070)	57 112	2.07 (3)	3	.69
9	Clutch APP	Face	MIL-L-7808 MIL-L-23699	P658RC USG 39	440C stainless	Fluoro-elastomer	5.05 (.7834)	95.5 (3.770)	8 200	2 460 (8 080)	31.2 (7)	6.16 (8.93)	1.65 (.065)	72 195	1.38 (2)		.58
10	Input shaft	Face	MIL-L-7808				8.06 (1.2504)	91.3 (3.595)	7 092	2 095 (6 870)	62.1 (14.2)	7.82 (11.36)	2.72 (.107)	78 043	0 (0)	High	1.22
11	Compressor	Face	MIL-L-7808	Carbon		Fluoro-elastomer	7.13 (1.1058)	90.6 (3.571)	16 000	4 560 (14 958)	66.7 (15)	9.35 (13.56)	2.29 (.090)	202 903	1.38 (2)	High	1.30

[a] The seal diameter is the primary ring nose inside diameter.

E-9298-1

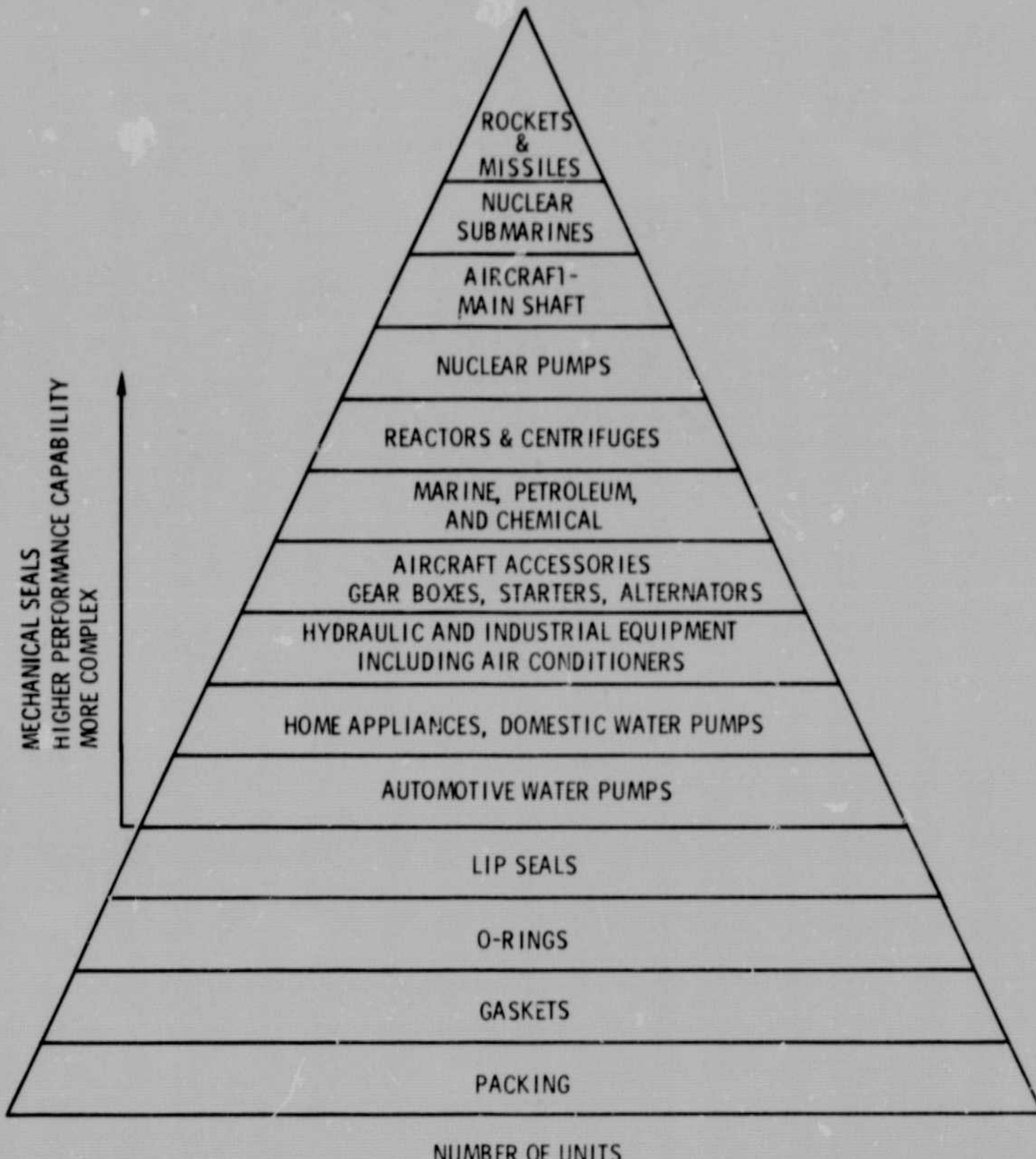


Figure 1. - Shaft seal usage.

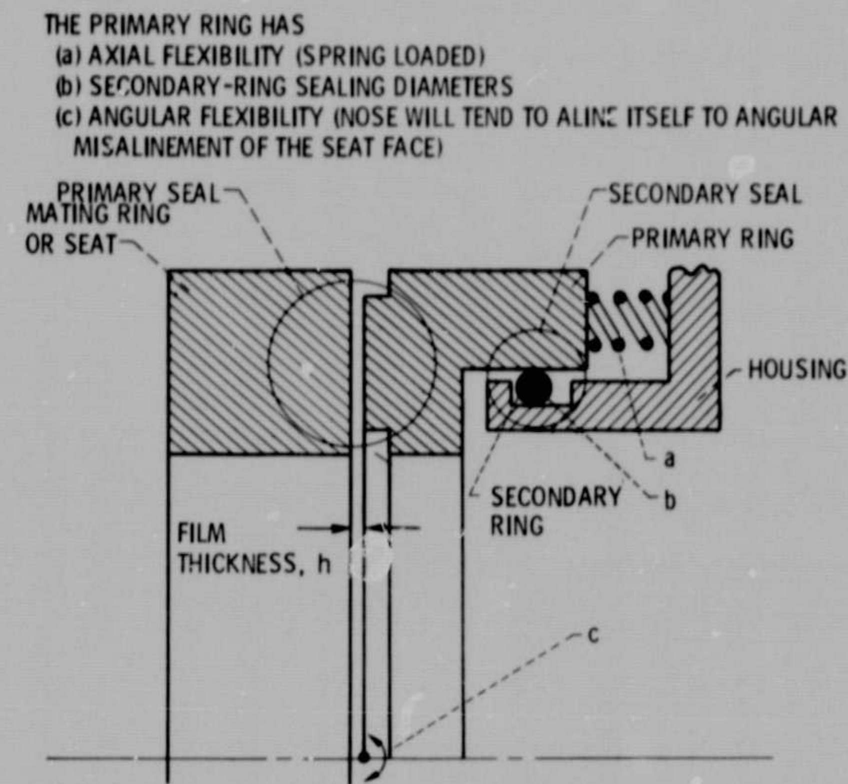


Figure 2. - Arrangement of seal components (nonrotating primary ring).

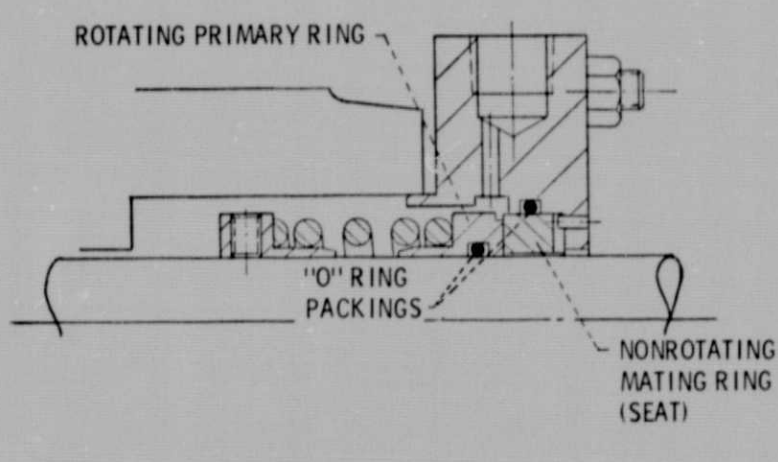
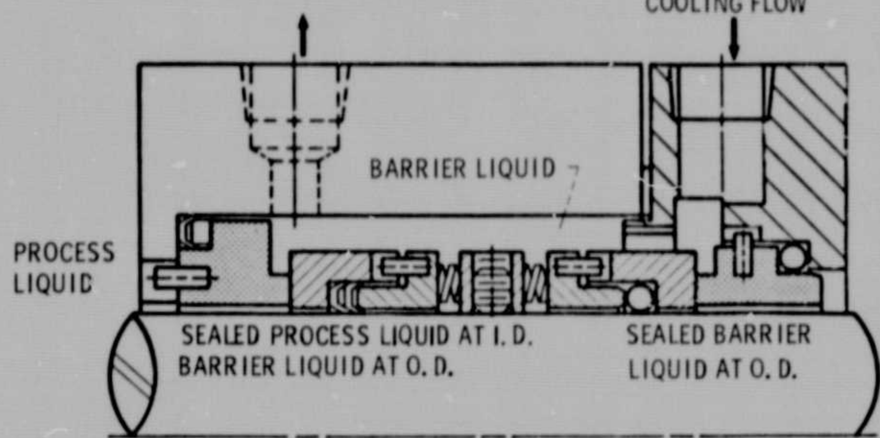
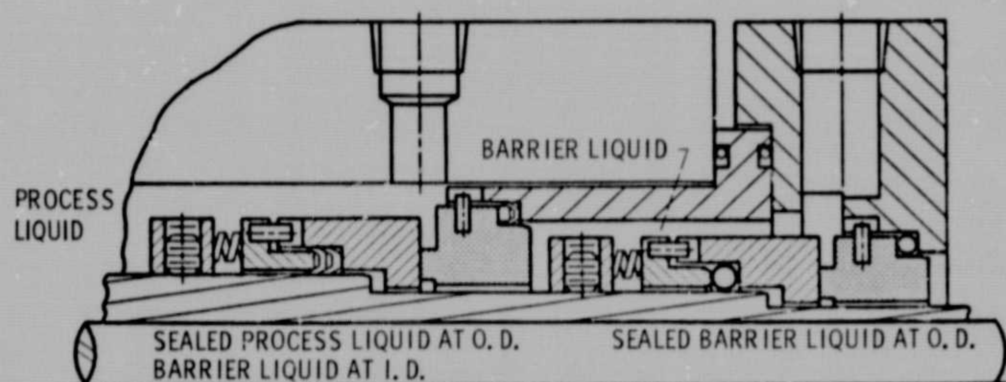


Figure 3. - Mechanical face seal with rotating primary ring.



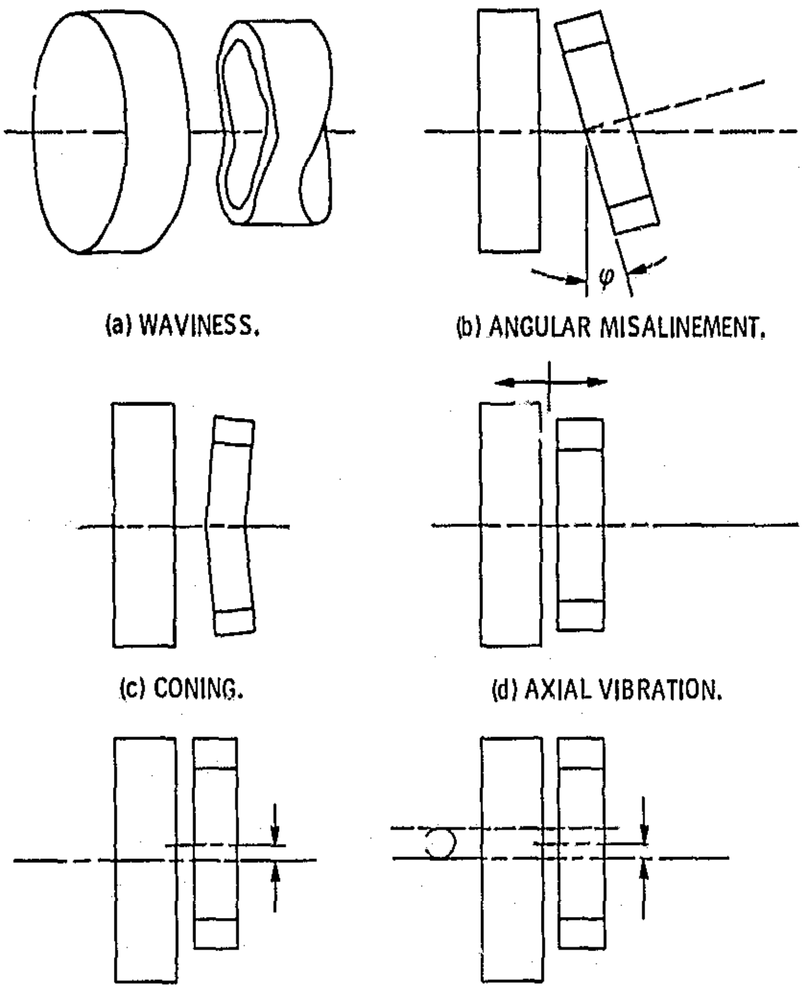
(a) DOUBLE SEAL.

E-9298-1



(b) TANDEM SEAL.

Figure 4. - Tandem seals, rotating primary-ring type.



(a) WAVINESS.

(b) ANGULAR MISALINEMENT.

(c) CONING.

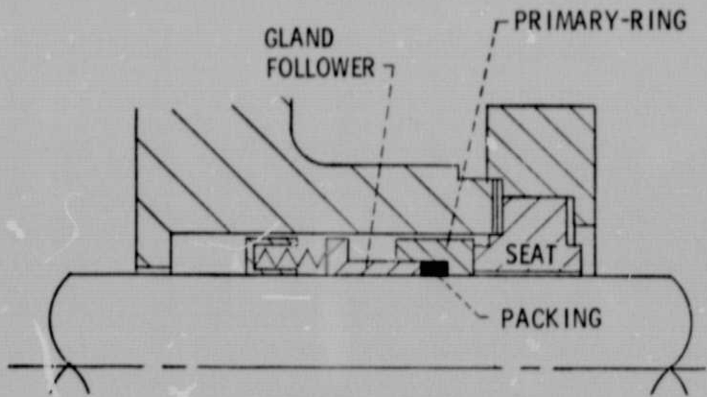
(d) AXIAL VIBRATION.

(e) PARALLEL MISALINEMENT.

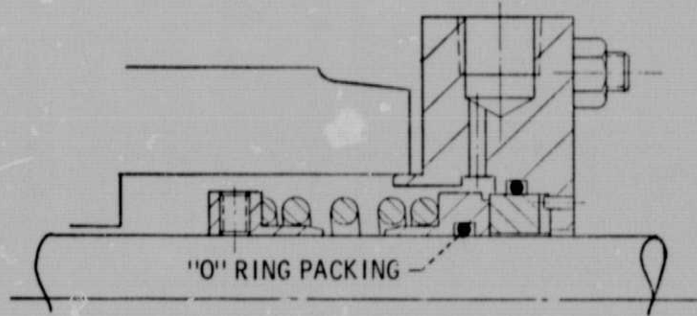
(f) SHAFT WHIRL.

Figure 5. - Possible primary-seal geometries.

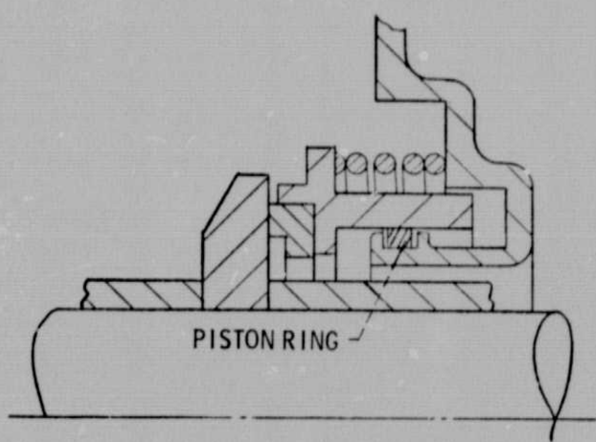
E-9298-1



(a) COMPRESSION PACKING.

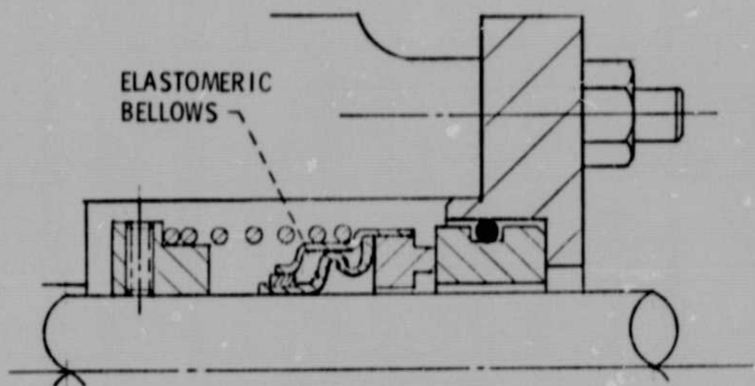


(b) AUTOMATIC PACKING, ELASTOMERIC.

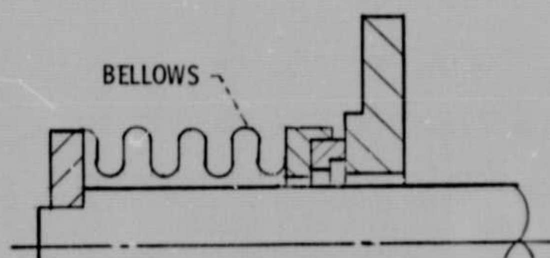


(c) AUTOMATIC PACKING, PISTON RING.

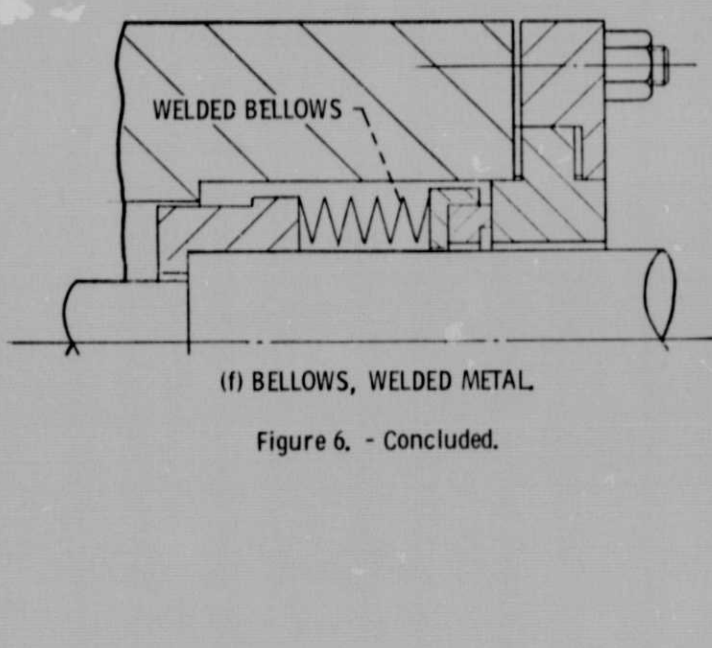
Figure 6. - Types of secondary seals.



(d) BELLOWS, ELASTOMERIC.



(e) BELLOWS, FORMED METAL.



(f) BELLOWS, WELDED METAL.

Figure 6. - Concluded.

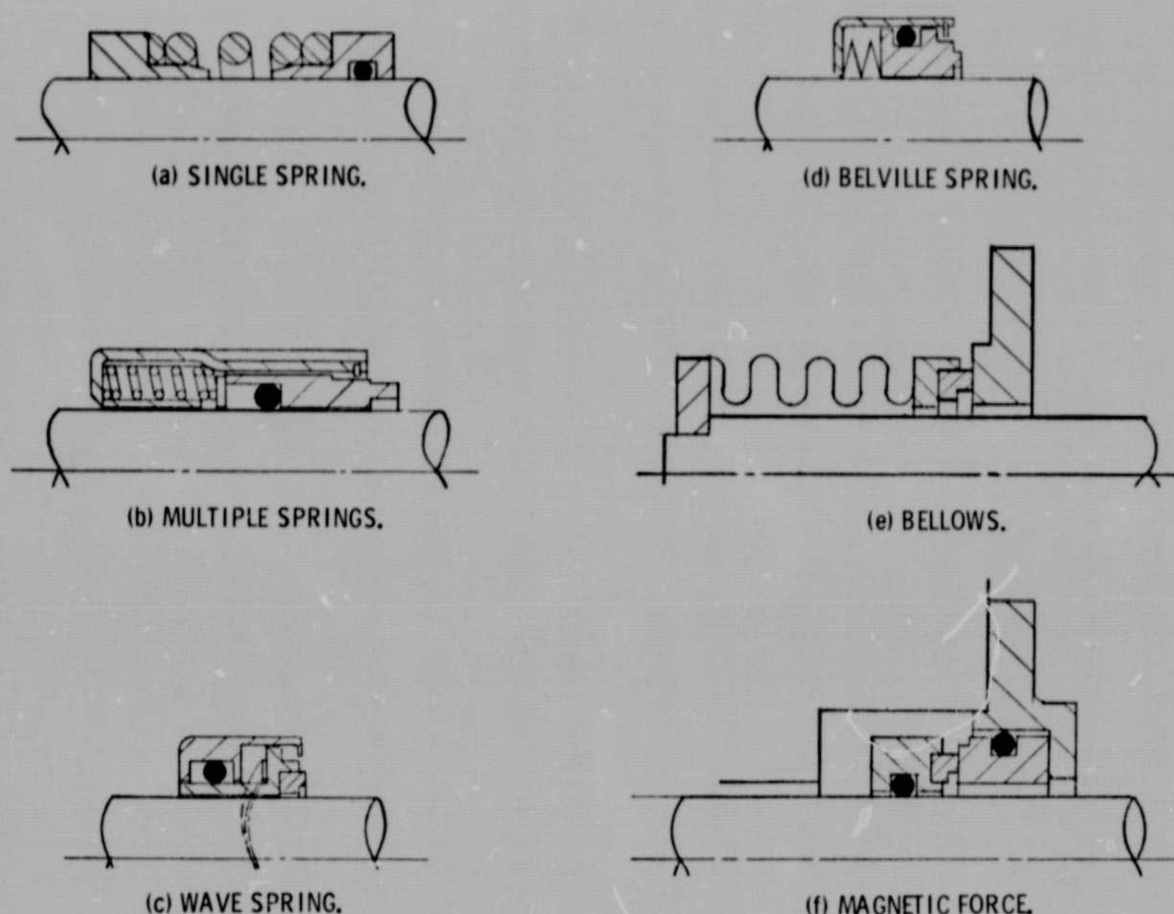


Figure 7. - Face loading elements.

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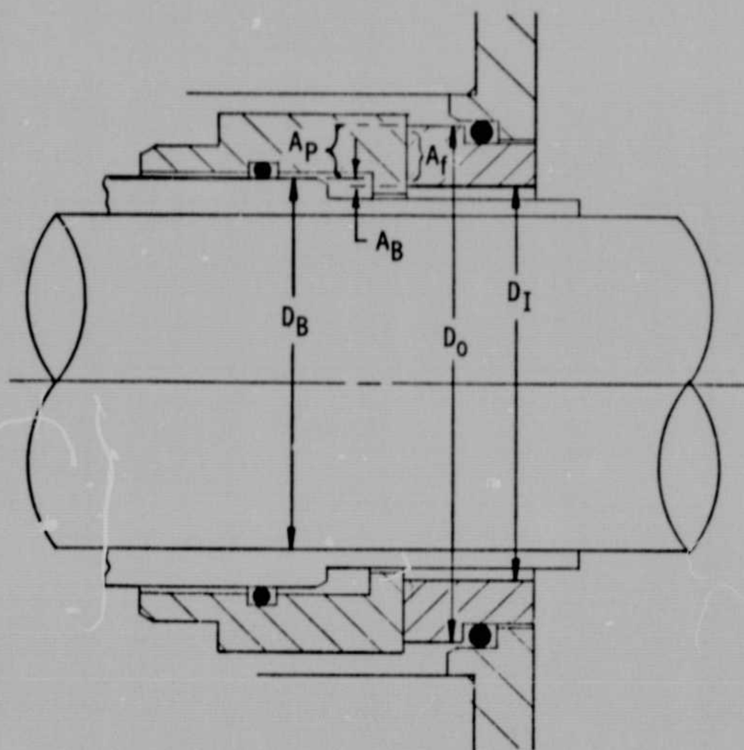


$$\text{PALANCE RATIO} = \frac{A_P}{A_f} = \frac{D_o^2 - D_B^2}{D_o^2 - D_I^2}$$

Figure 8. - Pressure balance for sealed pressure at the O. D.

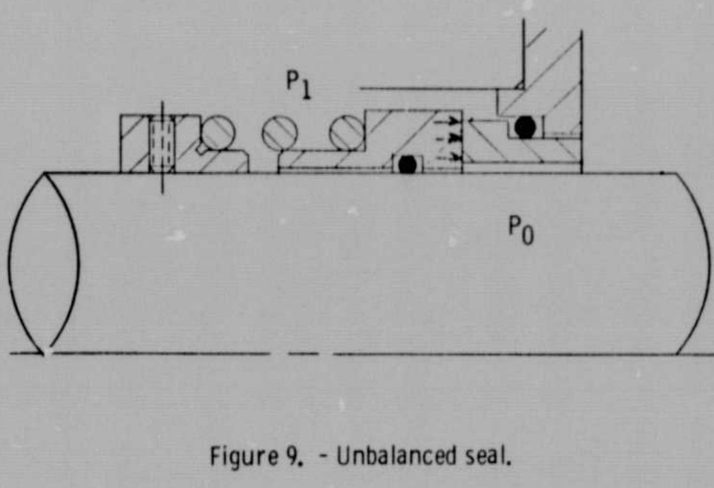


Figure 9. - Unbalanced seal.

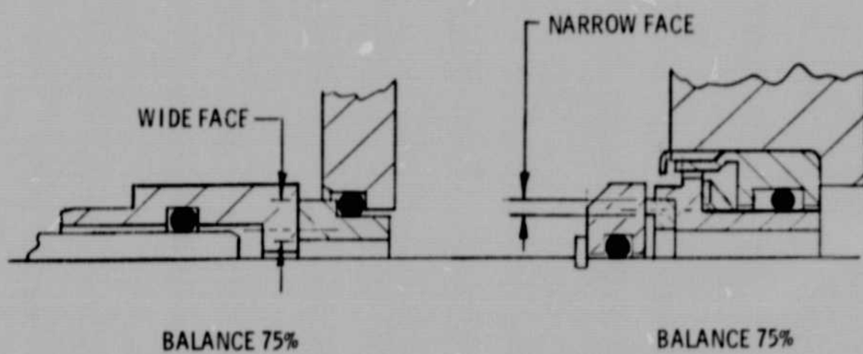
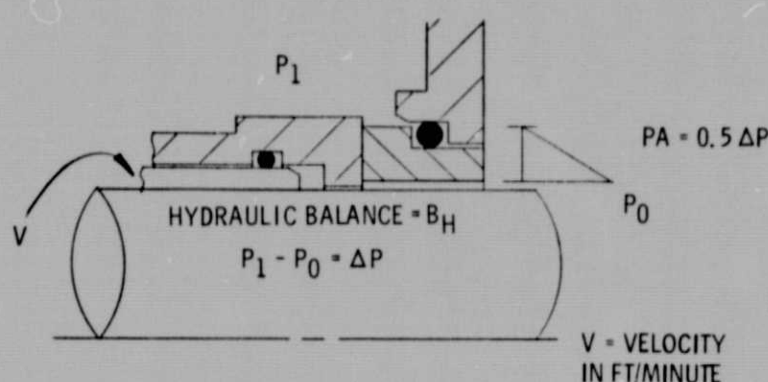


Figure 10. - Wide and narrow seal faces.

E-9298-1



(1) $PV = \Delta P \times V$

(2) $PV = B_H \times \Delta P \times V$

(3) $PV = (B_H \times \Delta P - 0.5\,\Delta P)V$

Figure 11. - Pressure balance formulas.

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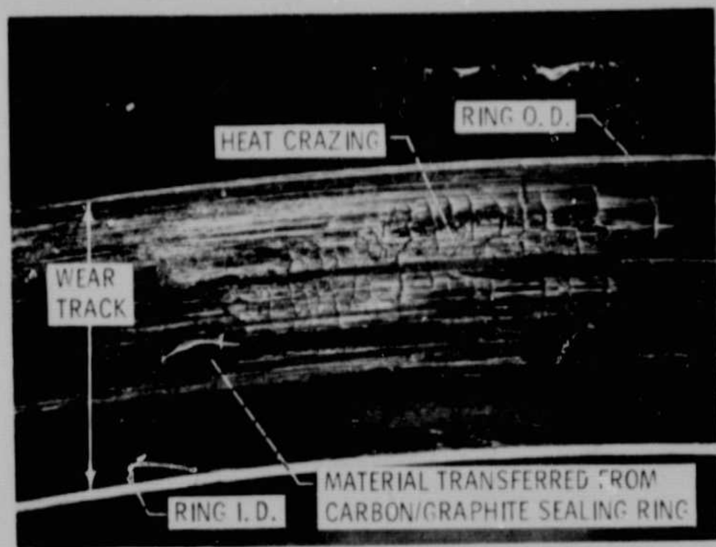

Figure 12. - Micrograph of mating ring sealing face. X12.

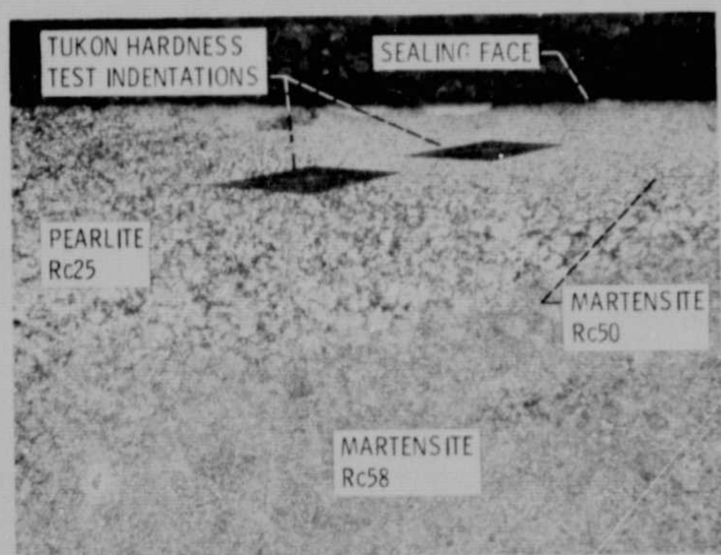

Figure 13. - Micrograph of mating ring cross-section. X200. "Hot spots" on sealing face are evidenced by localized annealing of ring material. Cooler portions of the sealing face were not annealed.

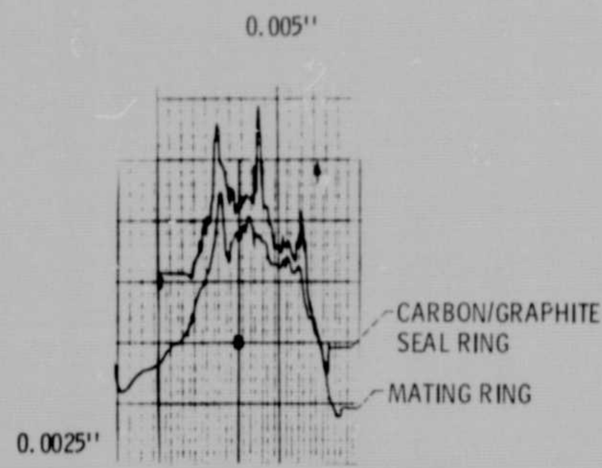

Figure 14. - Surface profile traces of mating seal members.

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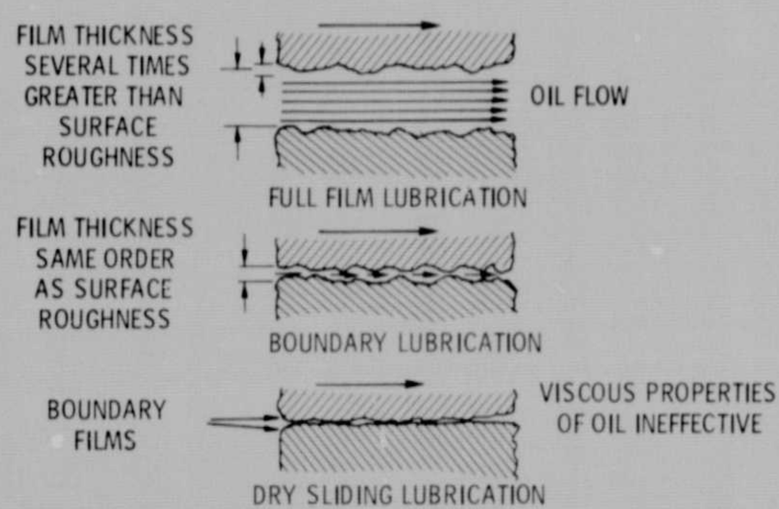

Figure 15. - Seal lubrication models.